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# A review of the applications of heat pipe heat exchangers for heat recovery

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ABSTRACT

The waste heat recovery by heat pipes is accepted as an excellent way of saving energy and preventing global warming. This paper is a literature review of the application of heat pipes heat exchangers for the heat recovery that is focused on the energy saving and the enhanced effectiveness of the conventional heat pipe (CHP), two-phase closed thermosyphon (TPCT) and oscillating heat pipe (OHP) heat exchangers. The relevant papers were allocated into three main categories, and the experimental studies were summarized. These research papers were analyzed to support future works. Finally, the parameters of effectiveness of the CHP, TPCT and OHP heat exchangers were described. This review article provides additional information for the design of heat pipe heat exchangers with optimum conditions in the heat recovery system.



**Contents**



## 1. Introduction

The use of heat pipes for waste heat recovery is an excellent way to save energy and prevent global warming. A heat pipe heat exchanger (HPHE) is utilized as an efficient air-to-air heat recovery device in both commercial and industrial applications. The HPHE is the best choice, with virtually no cross-leakage between the exhaust gas and supply air. It possesses many advantages, such as its heat recovery effectiveness, compactness, lack of moving parts, light weight, relative economy, small pressure drop on the air side, complete separation of hot and cold fluids, and reliability. The HPHE has been applied extensively in many industries (e.g., energy engineering, chemical engineering and metallurgical engineering) as waste heat recovery systems. One of the most important applications of HPHEs is the recovery of the heat from exhaust gases in a furnace stack. A comparison to a conventional furnace of the application of a heat pipe for heat recovery is illustrated in Fig. 1. In a conventional furnace (Fig. 1a), the exhaust gases enter directly into surrounding area, which not only wastes energy but also harms the environment. The use a HPHE (Fig. 1b) both reduces the primary energy consumption and protects the environment. However, research on the use of heat pipes for heat recovery, especially with regard to energy savings and environmental benefits, is needed.

The applications of the conventional heat pipe (CHP), two-phase closed thermosyphon (TPCT) and oscillating heat pipe (OHP) heat exchangers for heat recovery were reviewed. The results of this article provide additional information for the design of the heat pipe heat exchanger with optimum conditions and for the future research in this field.

## 2. Types of heat pipe heat exchangers

HPHEs are heat transfer devices in which the latent heat of vaporization is utilized to transfer heat over a long distance with a corresponding small temperature difference. It consists of closed tubes that are filled with a proper working fluid. The heat pipes can be classified as three types: a conventional heat pipe (CHP), a two-phase closed thermosyphon (TPCT) and an oscillating heat pipe

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**Nomenclature**

| | |
|---|---|
| ID | inner diameter (m) |
| OD | outer diameter (m) |
| FR | filling ratio |
| $W$ | wall thickness (mm) |
| $L$ | length of tube (m) |
| $Q$ | heat transfer rate (W) |
| $T$ | temperature (°C) |
| $V$ | velocity (m/s) |

*Subscripts*

| | |
|---|---|
| a | adiabatic |
| c | condenser |
| e | evaporator |
| in | input |
| out | output |

(OHP). In operation, as heat enters the evaporator, the equilibrium is perturbed and generates a vapor at a slightly higher pressure and temperature. The increased pressure causes vapor to flow along the pipe to the condenser section, where a slightly lower temperature causes the vapor to condense and release its latent heat of vaporization. The condensed fluid returns to the evaporator section through the capillary action of the wick in the CHP or the gravitational force in the TPCT.

TPCTs are essentially heat pipes, but without the wick structure. The difference between a CHP and a TPCT is that the TPCT uses gravity to transfer the heat from a heat source that is located below the cold sink. As a result, the evaporator section is situated below the condenser section. The working fluid evaporates, condenses in the condenser section and flows back to the evaporator section under the influence of gravity. When gravity can be utilized, TPCTs are preferred to heat pipes because the wicks in the heat pipes produce an additional resistance to the flow of condensate. Fig. 2 illustrates the principal difference between the CHP and the TPCT [1].

OHPs, or pulsating heat pipe (PHPs), are one of the latest developments in heat pipe technology. In contrast to a CHP, where the working fluid inside the heat pipe circulates continuously by capillary forces between the heat source and the heat sink in the form of a counter current flow, the working fluid in an OHP oscillates in its axial direction. The basic heat transfer mechanism in a pulsating heat pipe is the oscillating movement of the fluid associated with the phase change (evaporation and condensation) phenomena. The OHP is composed of a bundle of turns of one continuous capillary tube. The diameter of the capillary tube must be small enough to allow the liquid and vapor plugs to coexist. The basic principle of an OHP is that when one end of the bundle of turns of the undulating capillary tube is subjected to a high temperature, the working fluid inside evaporates and increases the vapor pressure, which causes the bubbles to grow in the evaporator zone. This pushes the liquid column toward the low temperature end (condenser). The condensation at the low temperature end will further increase the pressure difference between the two ends. Because of the interconnection of the tubes, the motion of the liquid slugs and vapor bubbles at one section of the tube toward the condenser also leads to the motion of slugs and bubbles in the next section toward the high-temperature end (the evaporator). Thus, the heat can be transferred from the heated section to the cooled section. The OHP has the advantage of not needing a wick structure to transport the liquid. There is also no pump, so the OHP is passive. To operate, it needs no power other than the heat that is being rejected. Although the overall resistance of an OHP is typically greater than that of a CHP, the OHP can operate with greater heat fluxes; the system utilizes boiling and is not

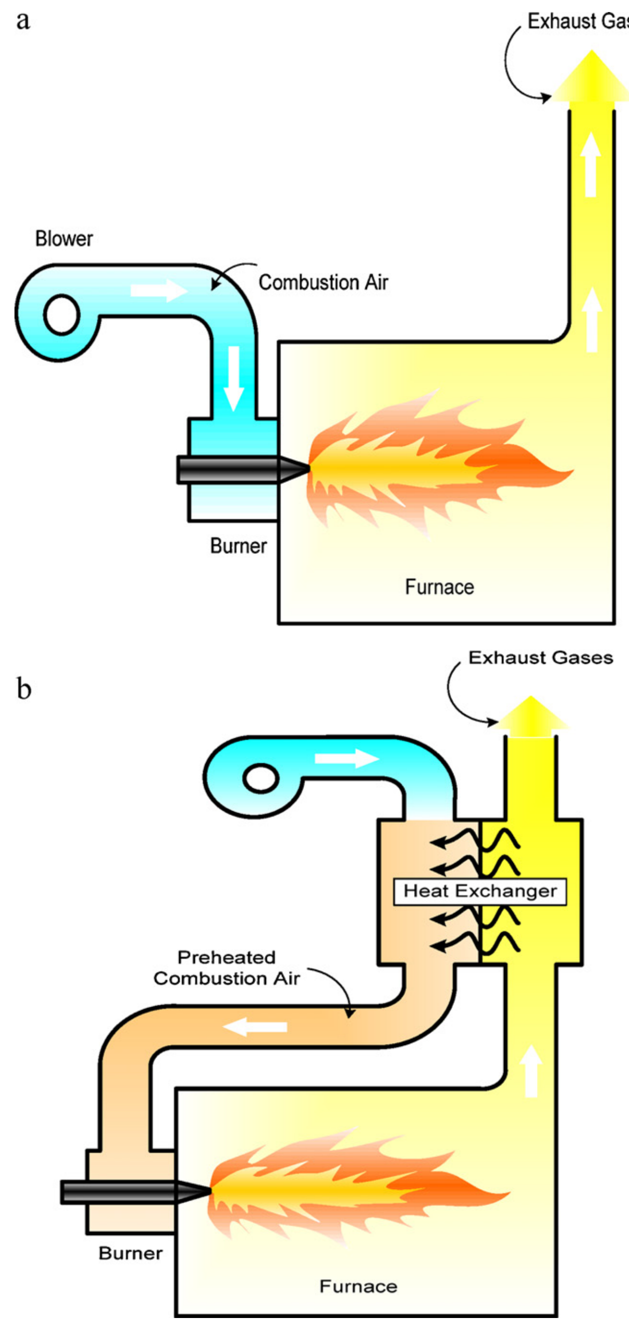


**Fig. 1.** Heat recovery for preheated air before combustion of the furnace.

limited by a boiling limit other that the critical heat flux. The HPHE is a self-contained, self-maintained passive energy recovery device. It has a very large coefficient of thermal transfer utilizing vapor liquid flows. The oscillation of the slugs and bubbles in the OHP is self-sustained by the evaporation of the liquid slugs and the condensation of the bubbles in pipes. The driving force is caused by the nucleate boiling and the condensate of working fluid. The HPHE can conduct heat from a high-temperature section to another section with a lower temperature. The OHP has the multiple advantages: inexpensive manufacturing, good heat-transfer performance, fast thermal response, operability in any position, and operational flexibility. The OHPs can be divided into three main types (Fig. 3). Fig. 3a is a closed-loop oscillating heat pipe (CLOHP), thus named because of its long capillary tube forming a closed loop. The heat is transferred by the oscillation of the working fluid in the direction of the longitudinal axis of the pipe with a superimposed bulk circulation

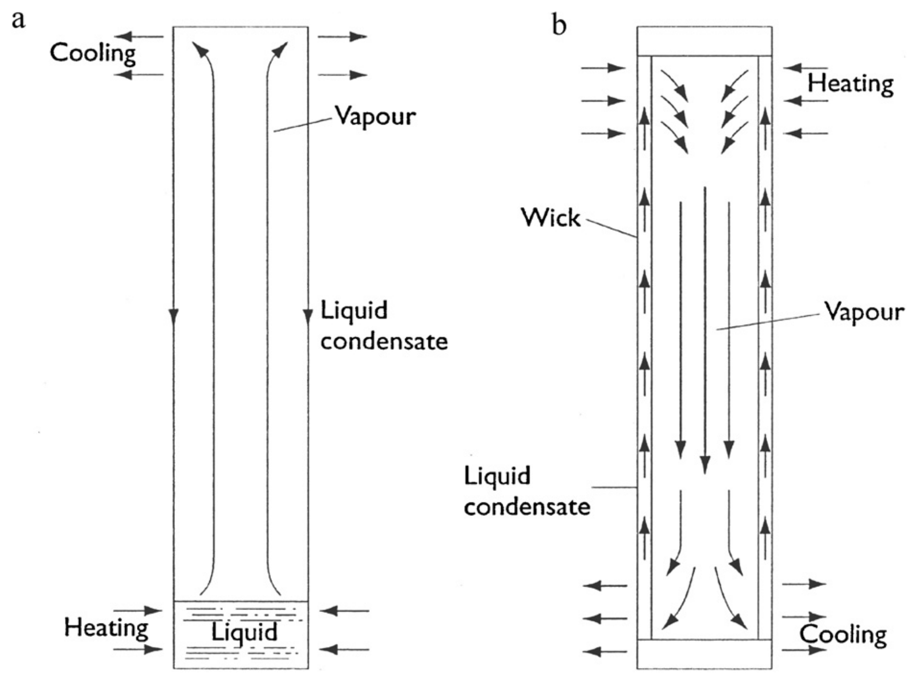


**Fig. 2.** Conventional heat pipe (CHP) and two phase closed thermosyphon (TPCT) [1].

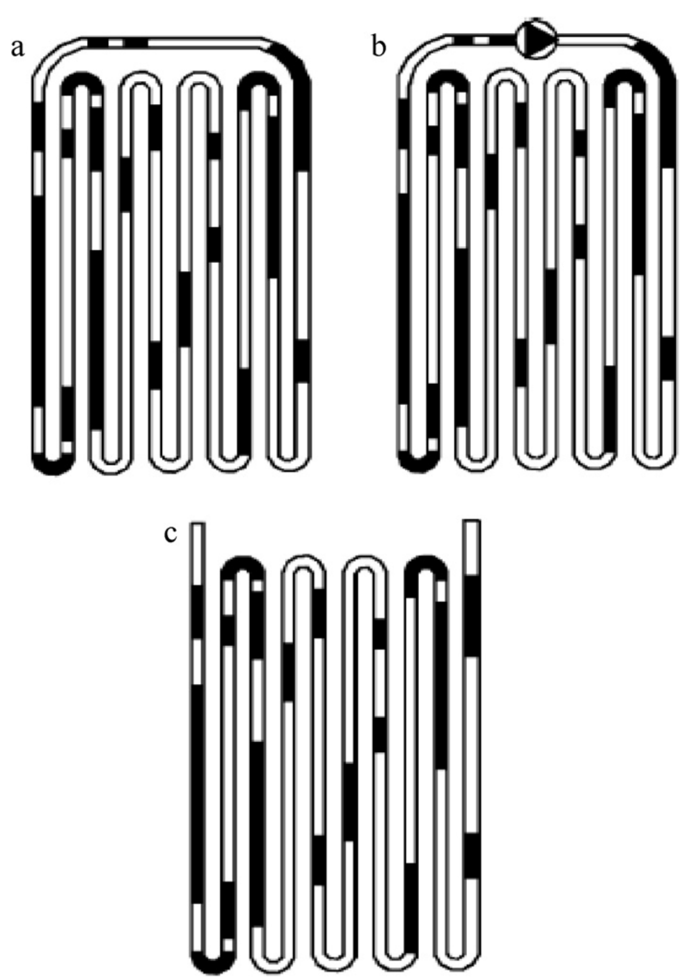


**Fig. 3.** Types of oscillating heat pipes.

in either direction. Fig. 3b is a closed-loop oscillating heat pipe with check valves (CLOHP/CVs). It is made from a long capillary tube with the ends joined to form a closed loop. The CLOHP/CVs incorporate one or more direction-control one-way check valves in the loop such that the working fluid can circulate in a specified direction only. Fig. 3c is a closed-end oscillating heat pipe (CEOHP) and is made from a long capillary tube, which is closed at both ends. In this case, the heat transfer occurs only through the oscillations that are driven by the rapid, fluctuating pressure wave perturbations.

## 3. Applications of heat pipe for heat recovery

Recently, researchers have produced developments of high interest in heat pipe technology for heat recovery. Studies have analyzed the application, design, construct, and thermal performance of heat pipes. Noie-Baghban and Majideian [2] examined the waste heat recovery using CHPs for surgery rooms in hospitals. The CHPs were designed for use with low-temperature sources (15–55 °C). The study found that the CHP effectiveness is 0.16, although it depended on the diameter and the fin gap. This value is very small because the CHP was designed for low-temperature operating conditions. Abd El-Baky and Mohamed [3] applied the CHP to the heat recovery between two streams of fresh and return air in an air conditioning system, where the incoming fresh air could be cooled. The ratios of the mass flow rate between the return and fresh air (1, 1.5, and 2.3) were tested to validate the heat transfer and the temperature change of the fresh air. During the tests, the fresh air inlet temperature was controlled in the range of 32–40 °C, while the inlet return air temperature was kept constant at approximately 26 °C. Martinez et al. [4] designed a mixed-energy recovery system consisting of two CHPs and indirect evaporative recuperators for the air conditioning. The energy characterization of the mixed-energy recovery system was performed with the experimental design techniques. A main conclusion was that by applying the mixed-energy recovery system in the air-air conditioning installations consisting of two CHPs and indirect evaporative systems, part of the energy from the return airflow could be recovered, thus improving the energy efficiency and reducing the environmental impact.

**Table 1**
Summarized the geometric characteristics of CHP, TPCT and OHP heat exchangers.

| | Pipe | Number of tubes | Working fluid/FR | Fins | Wick | Authors |
|---|---|---|---|---|---|---|
| CHP | Material: copper<br>OD: 15 mm<br>$W_t$: 3 mm<br>$L_e$: 300 mm, $L_t$: 600 mm,<br>$L_c$: 300 mm | 8 | Material: methanol | None | 100 mesh SS | Noie-Baghban and Majideian [2] |
| CHP | Material: copper<br>OD: 12.7 mm<br>$W_t$: 50 cm<br>$L_e$: 20 cm, $L_a$: 10 cm,<br>$L_c$: 20 cm | 25 | Material: R11, R123 | Type: continuous fin<br>Material: aluminum<br>Thickness: 0.5 mm | 100 mesh brass | Abd El-Baky and Mohamed [3] |
| CHP | Material: –<br>OD: 12.7 mm<br>$W_t$: 2.1 mm<br>$L_t$: 62 cm | 12 | Material: 3.04 g of ammonia | None | 350 mesh Stainless steel | Martinez et al. [4] |
| TPCT Case 1 | Material: copper<br>OD: 15.88 mm<br>$W_t$: 1.22 mm<br>$L_e$: 300 mm, $L_a$: 150 mm,<br>$L_c$: 300 mm | 24 | Material: water<br>FR: 60% of the evaporator section | Type: continuous fin<br>Material: copper fin at evaporator, aluminum at condenser<br>Spacing: 472 fins per meter<br>Thickness: 0.162 mm | None | Lukitobudi et al. [5] |
| TPCT Case 2 | Material: steel<br>OD: 26.27 mm<br>$W_t$: 7.65 mm<br>$L_e$: 300 mm, $L_a$: 150 mm,<br>$L_c$: 300 mm | 10 | Material: water<br>FR: 60% of the evaporator section | Type: circular spiral fin<br>Material: steel<br>Spacing: 315 fins per meter<br>Thickness: 0.8 mm<br>Diameter: 52.7 mm. | None | |
| TPCT Case 3 | Material: copper<br>OD: 15.88 mm<br>$W_t$: 1.22 mm<br>$L_e$: 300 mm, $L_a$: 150 mm,<br>$L_c$: 300 mm | 24 | Material: water<br>FR: 60% of the evaporator section | None | None | |
| TPCT | Material: steel<br>OD: 20 mm<br>$W_t$: 1.5 mm<br>$L_e$: 150 mm, $L_a$: 5 mm,<br>$L_c$: 150 mm | 50 | Material: water<br>FR: 35% of the evaporator section | Type: plate fin<br>Material: steel<br>Spacing: 315 fins per meter<br>Thickness: 1.5 mm<br>Height: 8 mm. | None | Yang et al. [6] |
| TPCT Case1 | Material: copper<br>OD: 0.127 m<br>$W_t$: 1.5 mm<br>$L_t$: 0.45 m | 7 | Material: methanol<br>FR: – | Type: plate fin<br>Material: copper<br>Total: 70 fins<br>Thickness: 0.45 mm.<br>Height: 0.048 m. | None | Riffat and Gan [7] |
| TPCT Case2 | Material: copper<br>OD: 0.127 m<br>$W_t$: 1.5 mm<br>$L_t$: 0.45 m | 3 | Material: methanol<br>FR: – | Type: cylindrical spine fins<br>Material: copper<br>Total: 300 fins<br>Thickness: 0.45 mm.<br>Diameter: 0.7 mm. | None | |
| TPCT Case3 | Material: copper<br>ID: 18 mm<br>$L_t$: 365 m | 6 | Material: methanol<br>FR: – | Type: louvred fin<br>Material: aluminum<br>Total: 96 louvres<br>Spacing: 2 mm.<br>Height: 60 mm. | None | |
| TPCT | Material: copper<br>ID: 15 mm<br>$L_t$: 660 mm, $L_e$: 300 mm,<br>$L_c$: 300 mm | 24 | Material: R22<br>FR: 60% of the evaporator section | Type: plate fin<br>Material: aluminum<br>Total: 32 fins/100 mm.<br>Thickness: 0.164 mm.<br>Height: 140 mm. | None | Wu et al. [8] |
| CEOHP | ID: 2 mm<br>$L_e$: 190 mm<br>$L_c$: 190 mm<br>$L_t$: 600 mm | 1 | Material: water<br>FR: 50% of the evaporator section | – | None | Rittidech et al. [9] |
| CLOHP/CVs | ID: 2 mm<br>$L_e$: 190 mm<br>$L_a$: 8 mm<br>$L_c$: 190 mm<br>$L_t$: 358 mm | 1 | Material: R134a<br>FR: 50% of the evaporator section | – | None | Meena et al. [10] |

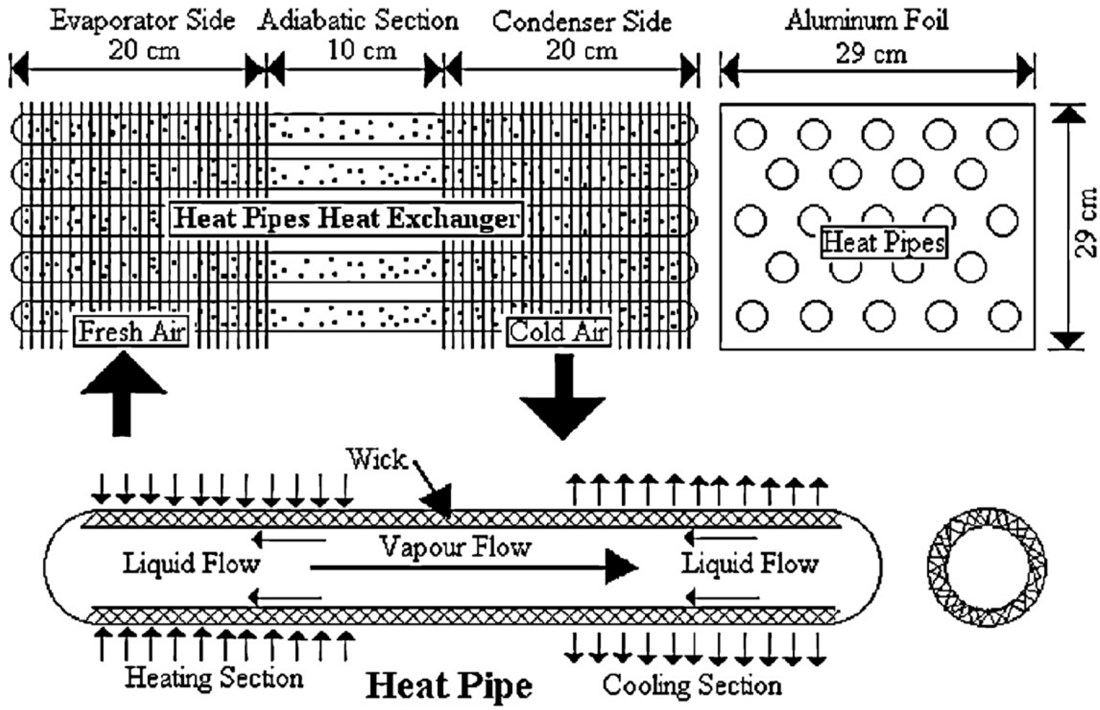


**Fig. 4.** The CHP heat exchanger for heat recovery [3].

In the applications of TPCT, Lukitobudi et al. [5] designed, constructed and tested a TPCT heat exchanger for a medium-temperature heat recovery in bakeries. The TPCT was very efficient (65%), although the authors commented that the overload pressure during the processes may damage the TPCT. Yang et al. [6] studied the possible application of a TPCT for the passengers in a large bus by recovering the heat from the exhaust gas of the engine. The study determined that TPCTs can be effectively used as a device for heat recovery, and the experimental results agree well with the numerical results. Riffat and Gan [7] explored the effectiveness of the TPCT heat exchangers for naturally ventilated buildings. In this research, the performances of three types of TPCT heat recovery units were tested in a two-zone chamber with a horizontal partition. The first TPCT heat exchanger consists of a bank of seven externally finned heat pipes, the second has a cylindrical spine fins, and the third was made of 2 rows of staggered TPCTs. CFD modeling was used for the pressure loss characteristics of the units. According to the experimental results, the air velocity significantly influenced the effectiveness of the TPCT heat recovery units. For the same velocity, the heat recovery was between 16% and 17% more efficient using two banks of TPCTs with plain fins instead of using one bank. Based on the CFD modeling results, at a velocity of 1 m/s, the predictive pressure loss coefficient for a 2-row, parallel setup of six pipes was 3.3 compared with 4.2 for the staggered pipes and 3.7 for seven smaller parallel TPCTs. The study recommended that in naturally ventilated low-rise buildings without the influence of wind, the designed mean air velocity should be less than 1 m/s. In another study, Wu et al. [8] studied the application of TPCT exchangers for the humidity control in air-conditioning systems. This type of heat exchanger can be an advantageous replacement for a conventional reheat-coil, resulting in energy savings and enhancing the cooling capability of the cooling coils with little or no external energy needed.

For the applications of OHP as a heat exchanger, Rittidech et al. [9] used a CEOHP air-preheater for the drying process. The method used connected the condenser section to the fresh-air section, and the evaporator section was in contact with the heat source from the gas burner. The dryer-bath setup was selected to be appropriate for the CEOHP air preheater. The heat-transfer rate and effectiveness were determined and compared with the predicted values. The study found that energy can be saved by using the CEOHP heat exchanger for air preheating in the drying system. In a similar study, Meena et al. [10] designed the CLOHP/CVs for the air preheater to reduce the relative humidity in the drying systems. The study demonstrated that using the CLOHP/CV air preheater can reduce the relative humidity in the drying system and can save energy.

## 4. Air-to-air heat pipe heat exchanger and its test rig

The geometric characteristics of the CHP [2–4], TPCT [5–8] and OHP [9,10] heat exchangers investigated in each study are summarized in Table 1. Based on the survey of previous research, the CHP, TPCT and OHP were designed with different forms for heat recovery (Figs. 4–7). The details of each paper will be described. In the experimental rig of [2] (Fig. 8a), the vertical CHP was installed between two zinc boxes with two fans that provide a flow rate of 0.103 $m^3$/s through the evaporator and condenser sections. The air velocities of both streams were measured by air velocity meters and were 2.3 m/s. The atmospheric air is heated by the three electric heating elements (total power 1500 W) for the lower box. After giving part of this heat to the evaporator section, the air is discharged to the atmosphere. In the experimental setup of [3] (Fig. 8b), the test section consists of two air ducts of 0.3 m × 0.22 m section areas connected by a horizontal CHP with fins. A square hole of 0.3 m × 0.3 m was made in one side of the two ducts for the installation of the CHP heat exchanger. The fresh-air duct was equipped with a blower to supply air to the evaporator side of the CHP heat exchanger. The return-cold and fresh-warm air ducts were insulated with glass wool (50 mm thick) to minimize the heat transfer to the surrounding air. The flow rates of air in both ducts were measured with a pitot-static tube. The fresh air was maintained at 0.4 kg/s, while the return air varied, with values of 0.4, 0.6 and 0.933 kg/s. Fig. 9 shows a schematic diagram of the experimental installation of [4] and the comprising systems comprised of the following

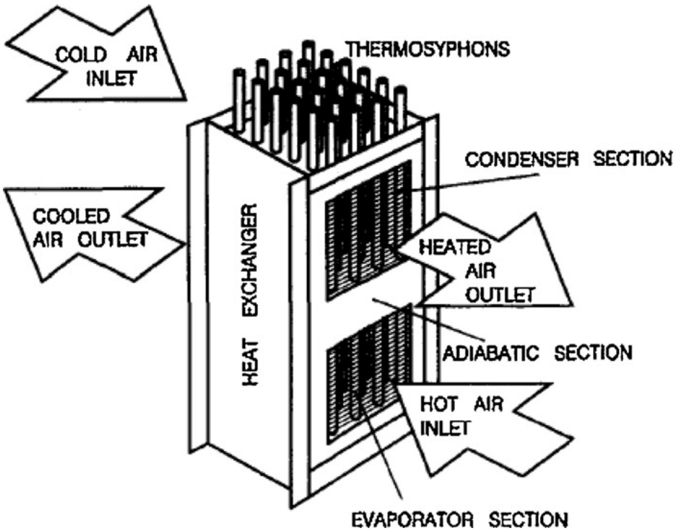


**Fig. 5.** The TPCT heat exchangers for heat recovery in a bakery [5].

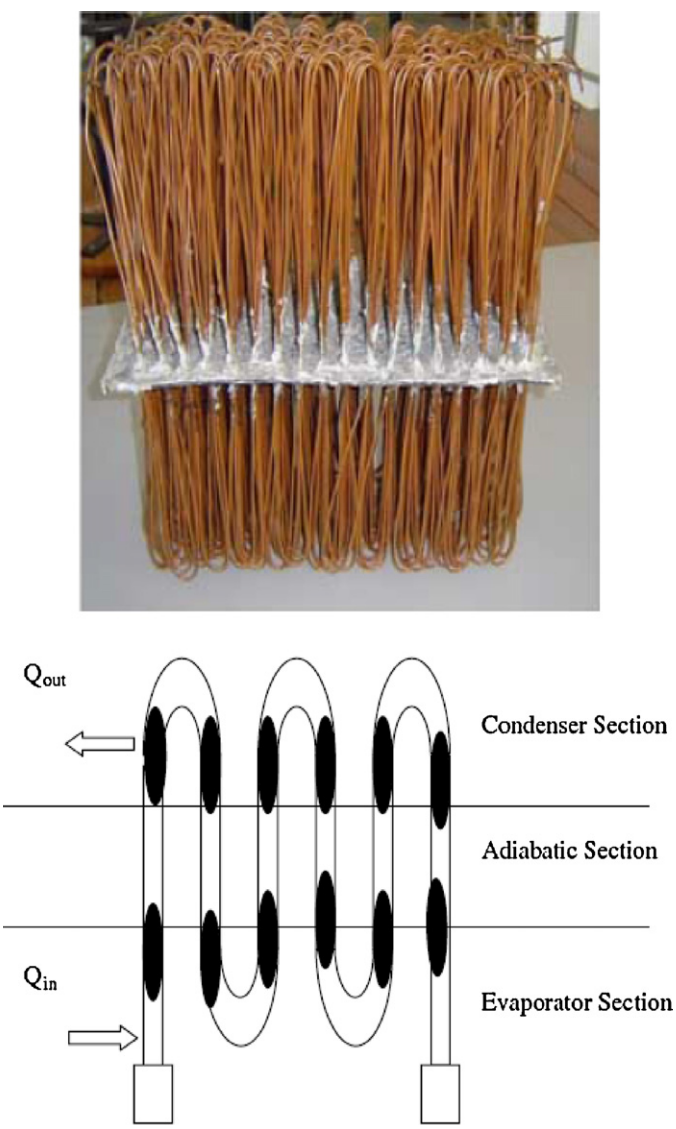


**Fig. 6.** The CEOHP air-preheater [9].

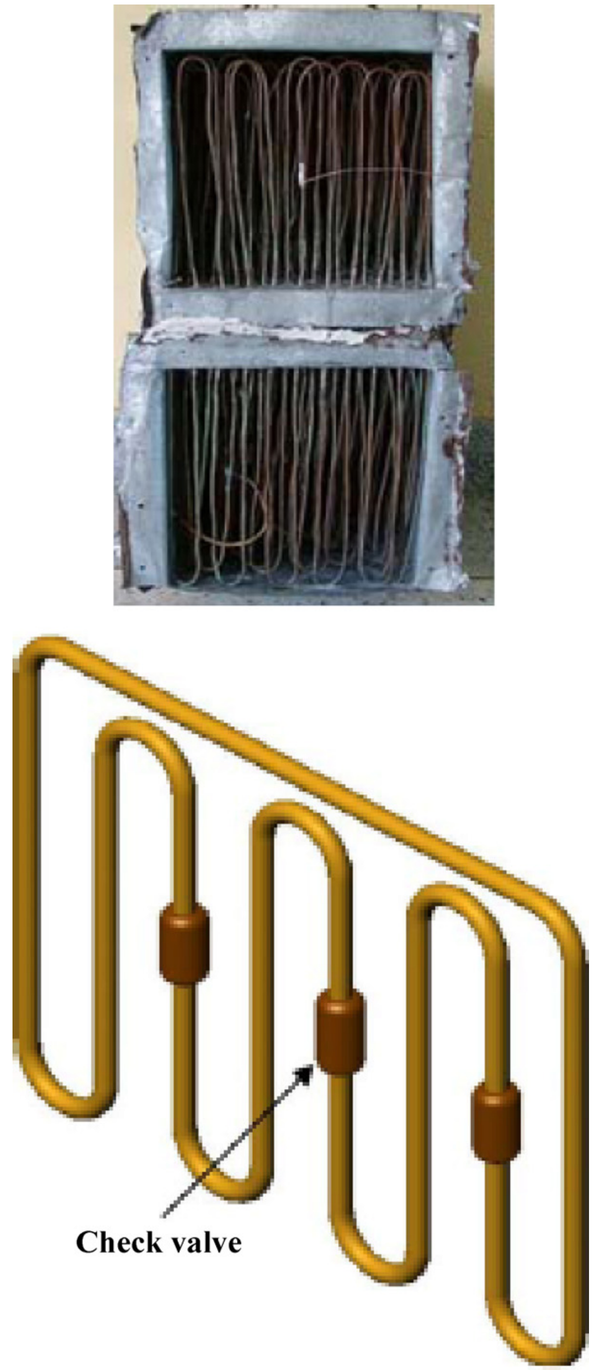


**Fig. 7.** The CLOHP/CVs for air-preheater [10].

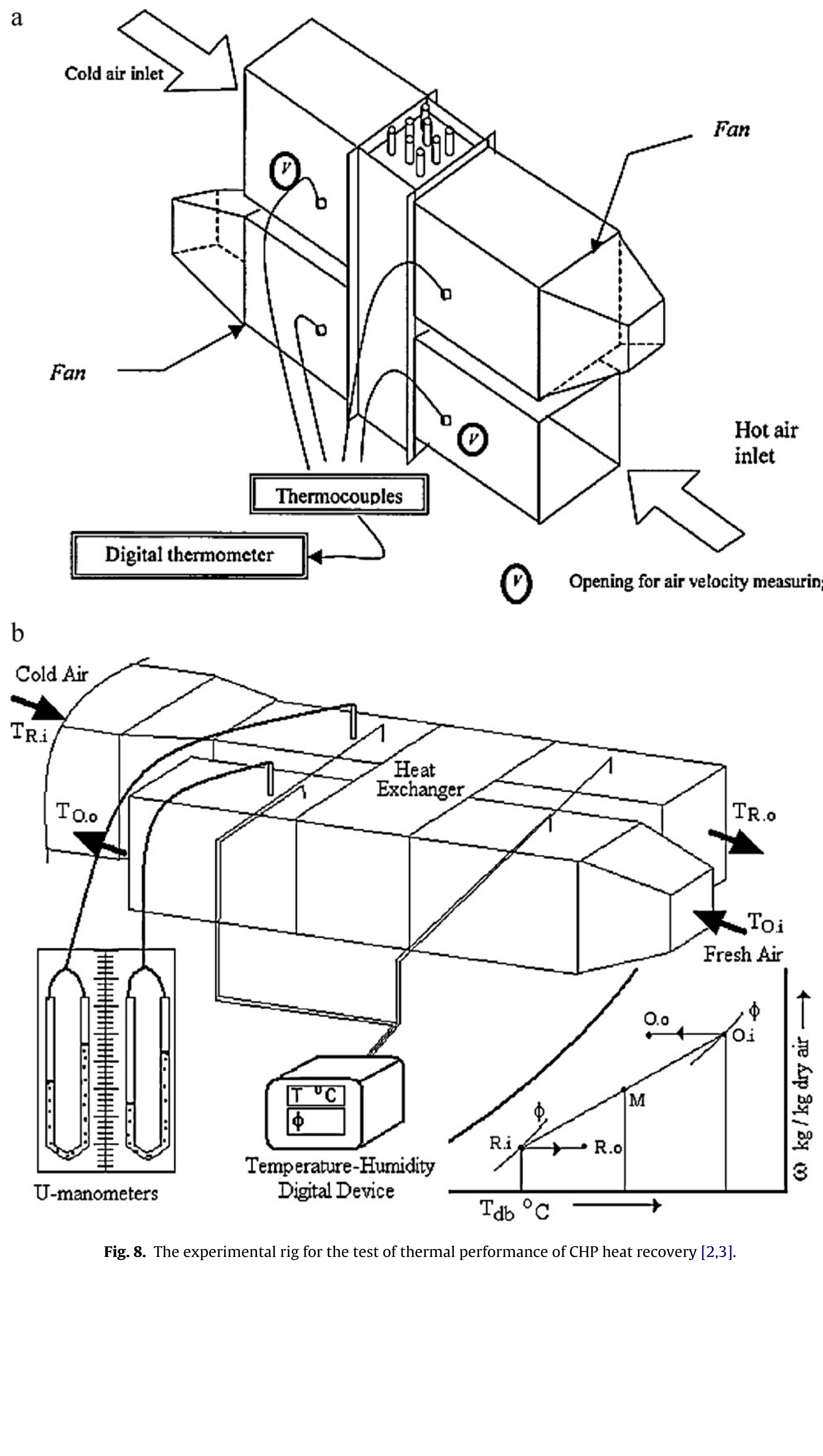


**Fig. 8.** The experimental rig for the test of thermal performance of CHP heat recovery [2,3].

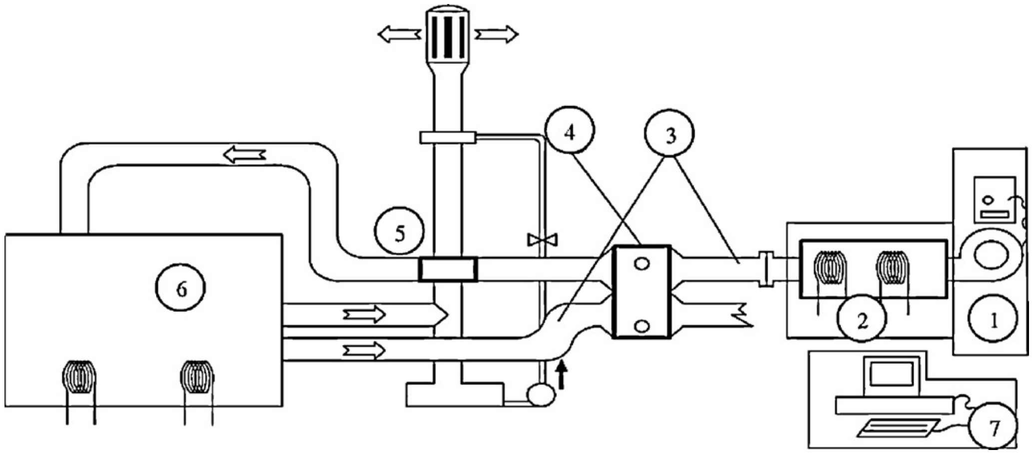


**Fig. 9.** The schematic diagram for the test of thermal performance horizontal CHP [4].

components: (1) the air handle unit with variable flow air, which simulates the outside air (temperature, humidity and flow air); (2) a room maintaining the supply parameters; (3) air distribution ducts; (4) heat pipe recuperator; (5) indirect evaporative recuperator; (6) a room (1.2 m × 1.2 m × 1.5 m) equipped with an air–air heat pump to maintain the comfort parameters; and (7) a data acquisition computer (hardware and software) for monitoring. The test rig of [5] was used to test the effectiveness of the TPCT (Fig. 10), which consists of air heated in the upper condenser section boosted by an electric heater and returned as the heating air to the evaporator section in a counter flow configuration. The fan is driven by a variable speed motor, and the air face velocity can vary between 1 and 5 m/s. The heat input into the evaporator section inlet can vary between 4 and 20 kW, depending on the number of activated

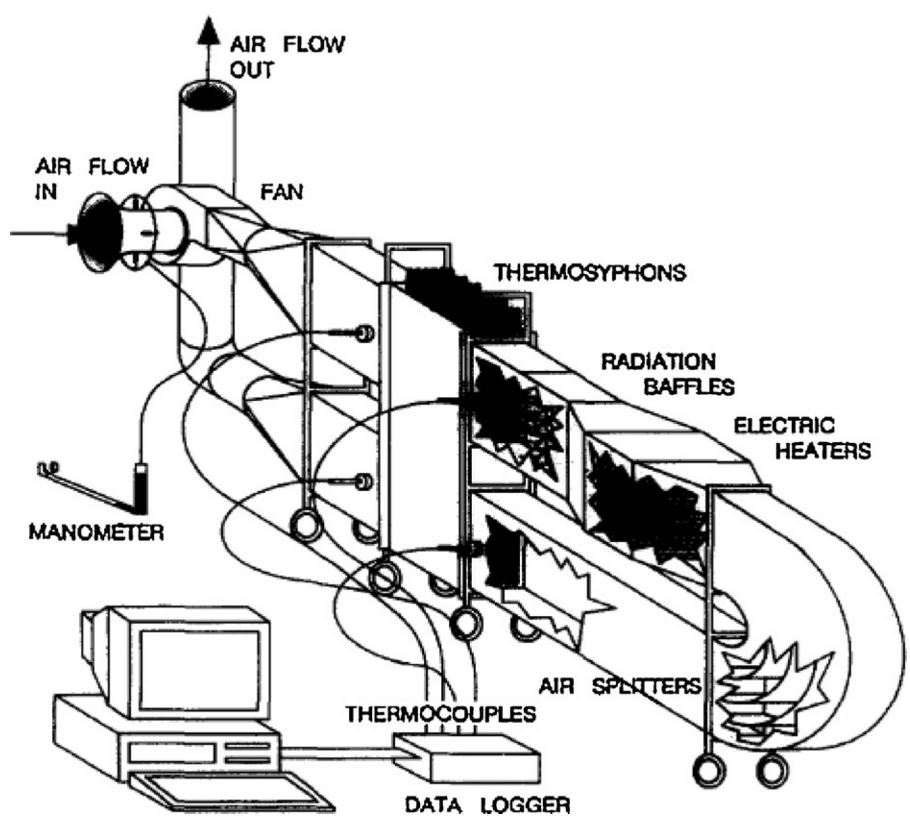


**Fig. 10.** The experimental set up used for the test of TPCT's effectiveness [5].

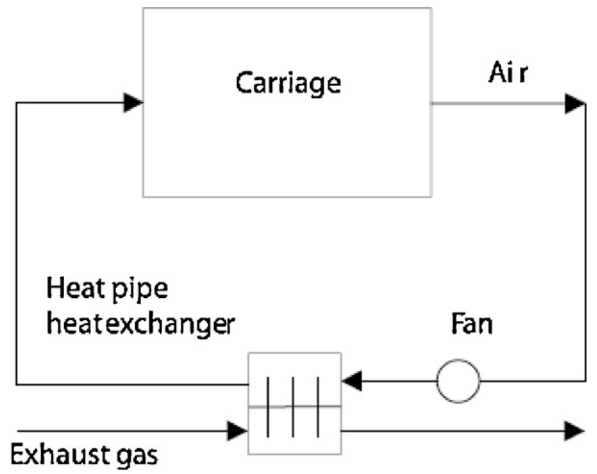


**Fig. 11.** The schematic diagram for warming passengers by recovery of the heat from exhaust [6].

heating elements. Thermocouples (type K) are used for the temperature monitoring. These measurements are processed by a computerized data logging system. Fig. 11 shows the experimental setup of [6], which consists of heat pipes in a staggered equilateral triangle arrangement of 20 mm. There are nine rows and approximately 50 tubes in this heat exchanger. The heat exchanger is manufactured with a cavity dimension of 210 mm × 305 mm × 350 mm. The steel pipe for filling the working fluid water has an outer diameter of 20 mm, pipe thickness of 1.5 mm, pipe length of 310 mm, evaporator section length of 150 mm, and a preformed assembled length for the end of the condenser section of 5 mm. The external ring steel fin is adopted at the condenser section. The thickness and height of the fins are 1.5 and 8 mm, respectively. The spacing interval between the fins is 8 mm. Naked pipe is used in the heat pipe evaporator section to match the heat resistance. To investigate whether the heat exchanger meets the requirements, the measurements of the exchanger performance are necessary. The hot fluid channel inlet of the heat exchanger is connected with the muffler outlet of the bus. A fan with a 100 mm $H_2O$ pressure head, 400 $m^3$/h volume flow rate and 120 W power supply is adopted to blow the air over the heat exchanger. The ambient temperature is 8 °C. The physical dimensions of the driver room and the carriage inlet are 1840 mm × 2300 mm × 1620 mm and 1810 mm × 2300 mm × 7040 mm, respectively. The carriage is an integral steel structure with a passenger gate. The maximum rotational speed of the engine, which is a petrol motor with six air cylinders, is 3000 rpm, and the cylinder volume is 5.42 l. In another study, Riffat and Gan [7] assessed the performance of the TPCT heat recovery units for naturally ventilated buildings. The effectiveness of three heat-pipe units was measured in a two-zone chamber with

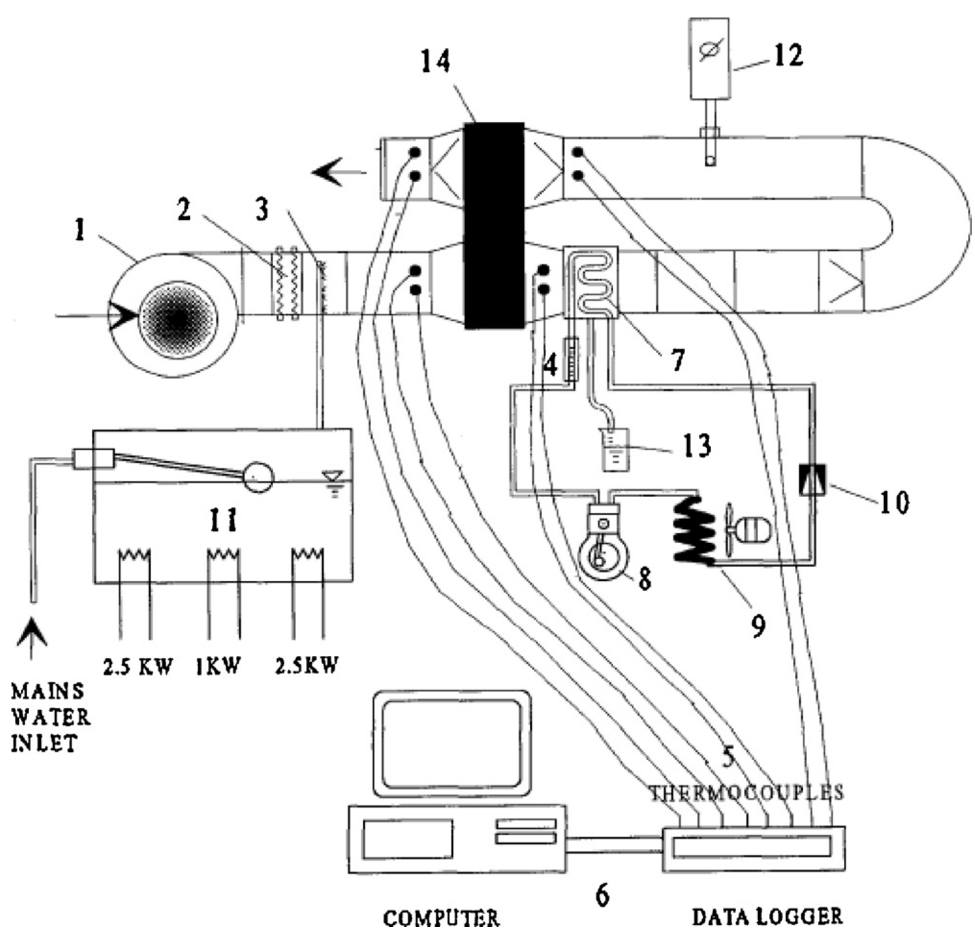


**Fig. 12.** Test rig for enchantment of the cooling capability [8].

an opening in a horizontal partition. The chamber was made of plywood insulated with a layer of polyurethane. The chamber had a net interior base area of 1.169 m × 1.133 m and a total height of 2335 m. It was divided into two zones by a horizontal partition. There was an opening (0.215 m × 0.215 m) in the middle of the partition to allow air to flow from one zone to another. The net internal volume of the chamber was 3.09 m$^3$. The supply and exhaust ducts were connected to the chamber on one of the vertical walls. The air ducts were also made of plywood. In the experimental rig of [8] (Fig. 12), intake air passing through the evaporator section of the HPHE was heated by an electric heater and humidified by steam, which was generated in an electric boiler. The precooled air leaving the evaporator of the HPHE then passed through a cooling coil for further cooling and was returned to the condenser section of the HPHE. The air stream flowed in a counterflow configuration. A variable-speed fan blew air through the ducts, which started as a 25 cm square section at the lower duct section and converted to a 25.4 cm diameter round duct at the return section. The test of [9] comprises the CEOHP air preheating for the drying process (Fig. 13). The tube arrangement is aligned in the direction of the hot-gas flow. The condenser section is connected to the fresh-air section, and the evaporator section is in contact with the heat source from the gas burner. The dryer-bath type is selected to be appropriate for the CEOHP air-preheater. The prototype duct, including the fiber insulation, has a cross-sectional area of 200 mm × 200 mm. The hot gas coming from the gas burner flows through the CEOHP air-preheater. In the experimental setup of [10] (Fig. 14), the hot air coming from the heater flows through the CLOHP/CV air-preheater. The temperatures at the inlet and outlet of the evaporator and the condenser section were recorded for the calculation of the heat-transfer rate and effectiveness.

These studies provide useful data for designers and researchers selecting a test rig to investigate the thermal performance of the HPHE heat exchangers for heat recovery systems.

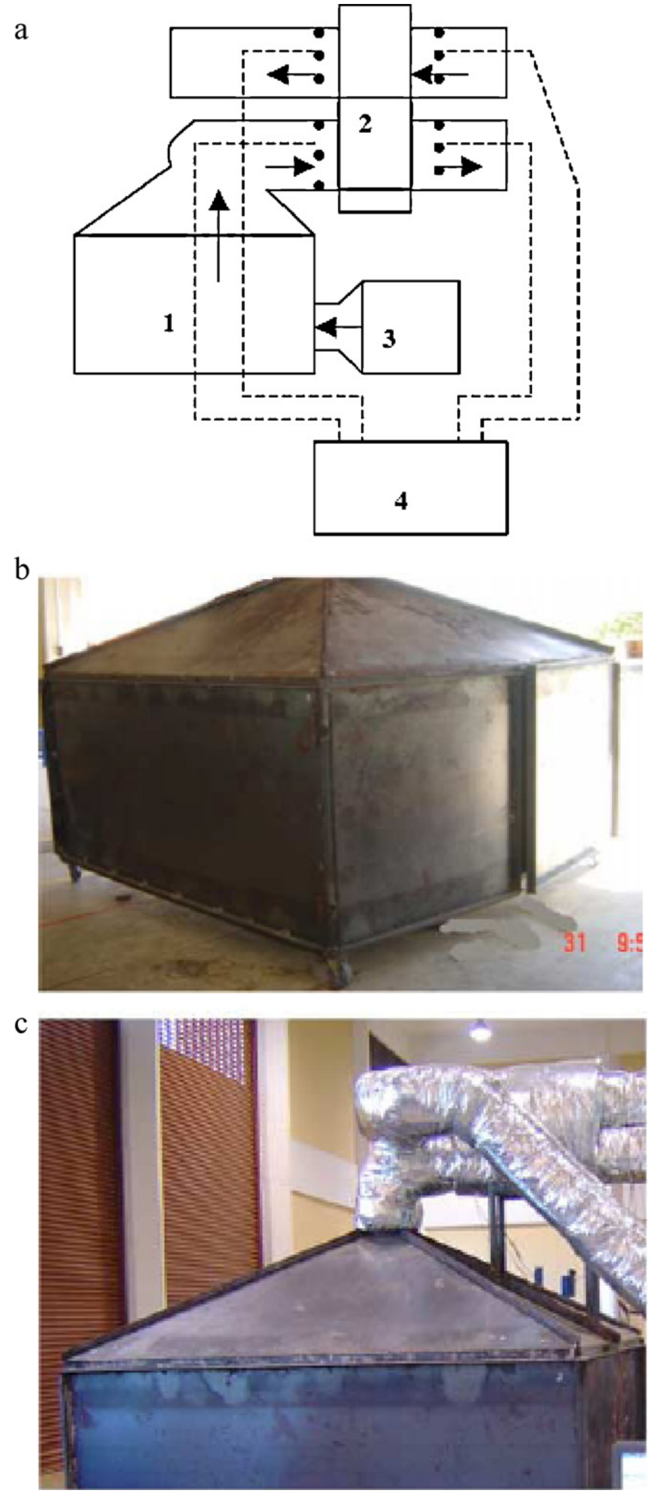


**Fig. 13.** The CEOHP air-preheater for energy thrift in a dryer [9].

## 5. Measuring the effectiveness of air-to-air heat pipe heat exchangers

This section highlights the measurements of the effectiveness of HPHEs used for heat recovery. The various parameters for the effectiveness of the previous research [2–10] are compared in Table 2. After the parameters were considered based on the effectiveness of air-to-air HPHEs, we conclude that the effectiveness depends on four main factors:

(1) The inlet temperatures in the evaporator section affect the overall effectiveness. This effect is reported by [2,6,9,10] (Fig. 15). The effectiveness increases with the inlet temperature.
(2) The hot and cold air velocities flow through the evaporator and condenser sections, respectively. The effectiveness decreases with an increase in the hot/cold air velocities (Fig. 16). This effect is reported by [3,5,7,8,10].
(3) In the geometric fin, which is the arrangement of the tubes, the in-line or staggered tube banks and the tube diameters influence the effectiveness. These effects are reported by [5,7] (Fig. 17).
(4) The working fluids influence the effectiveness of the HPHE. This effect is reported by [2,9]. If the working fluid changes from water to R123, the effectiveness increases because the R123 has a smaller latent heat (Fig. 15a).

**Table 2**
Comparison of the studied parameters for CHP, TPCT and OHP heat exchangers.

| Type | Control parameters | Variable parameters | Effectiveness | Authors |
|---|---|---|---|---|
| CHP | - Cold air, $V_{in} = 2.3$ m/s<br>- Hot air, $V_{in} = 2.3$ m/s | - Types of wick<br>- Rate of heat in put, 20–400 W | 0.16 | Noie-Baghban and Majideian [2] |
| CHP | - Return cold air Temperature, $T = 26\,^{\circ}$C | - Cold air, $T = 32$–$40\,^{\circ}$C<br>- Mass flow ratios, 1–2.33 | 0.26 | Abd El-Baky and Mohamed [3] |
| CHP | - Geometric heat pipe | - Primary air flow, 50–160 $m^3$/h<br>- Primary air temperature, 25–45 °C<br>- Primary air relative humidity (%), 30–60<br>- Water flow, 100–300 l/h<br>- Secondary air recirculation ratio (%), 25–75 | 82.5 | Martinez et al. [4] |
| TPCT | - Cold air, $T = 20\,^{\circ}$C<br>- Filling ratio<br>- Working fluid | - Hot and cold air, $0.5 \leq V \leq 5.5$ m/s<br>- Material of tubes<br>- Material and type of fins | 0.18–0.63 | Lukitobudi et al. [5] |
| TPCT | - Cold air velocity<br>- Filling ratio, 35% of evaporator section | - Exhaust gas temperature, $T = 100$–$300\,^{\circ}$C | 0.28 | Yang et al. [6] |
| TPCT | - Working fluid | - Different tube diameters and fins<br>- Air flow rate, 0.0145–0.1887 $m^3$/s | 0.22–0.64 | Riffat and Gan [7] |
| TPCT | - Working fluid<br>- Filling ratio | - Air face velocity, $V = 0.8$–$1.4$ m/s | – | Wu et al. [8] |
| TPCT | - Cold air, $T = 20\,^{\circ}$C<br>- Filling ratio<br>- Working fluid | - Hot and cold air, $0.5 \leq V \leq 5.5$ m/s<br>- Material of tubes<br>- Material and type of fins | 0.18–0.63 | Lukitobudi et al. [5] |
| TPCT | - Cold air velocity<br>- Filling ratio, 35% of evaporator section | - Exhaust gas temperature, $T = 100$–$300\,^{\circ}$C | 0.28 | Yang et al. [6] |
| TPCT | - Working fluid | - Different tube diameters and fins<br>- Air flow rate, 0.0145–0.1887 $m^3$/s | 0.22–0.64 | Riffat and Gan [7] |
| TPCT | - Working fluid<br>- Filling ratio | - Air face velocity, $V = 0.8$–$1.4$ m/s | – | Wu et al. [8] |
| CEOHP | - Hot gas, $V = 3.3$ m/s<br>- Filling ratio | - Working fluid<br>- Hot gas, $T = 60$, 70 or 80 °C | 0.38–0.54 | Rittidech et al. [9] |
| CLOHP/CVs | - Inner diameter tube<br>- Total length of tube | - Hot air, $V = 0.5$, 0.75 or 1 m/s<br>- Hot air, $T = 50$, 60 or 70 °C | 0.29–0.76 | Meena et al. [10] |

**Fig. 14.** The experimental set up for a reducing relative humidity in drying systems [10].

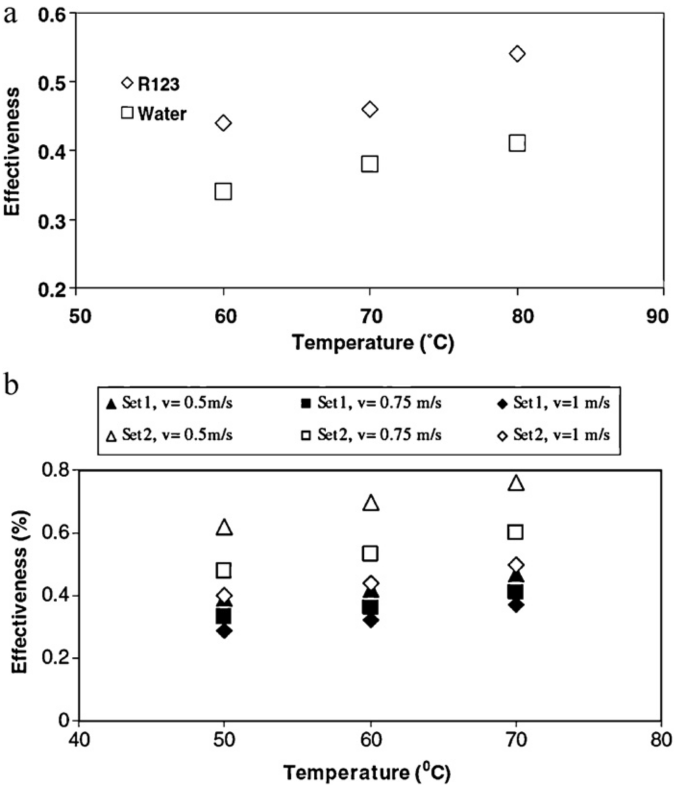


**Fig. 15.** Effect of inlet temperatures in evaporator on effectiveness [9,10].

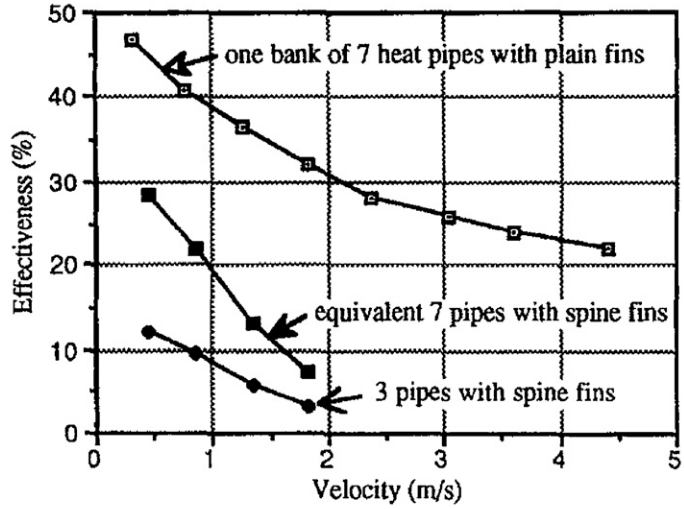


**Fig. 17.** Effect of tube bank and fin shape on effectiveness [7].

## 6. Conclusions

A literature review was performed on the applications of heat pipes heat exchangers for heat recovery. The unit of heat pipes for heat recovery contains an efficient air-to-air heat recovery device that controls heat loss in both commercial and industrial applications to ensure both energy savings and environmental protection. It does not need input power for its operation and does not require cooling water and lubrication systems. It is the best choice, with practically no cross-leakage between the exhaust air and supply air. The heat exchangers with the heat pipe units possess many advantages, such as heat recovery effectiveness, compactness, lack of parts, light weight, relative economy, smaller pressure drop of fluid flow across, complete separation of hot and cold fluids, and

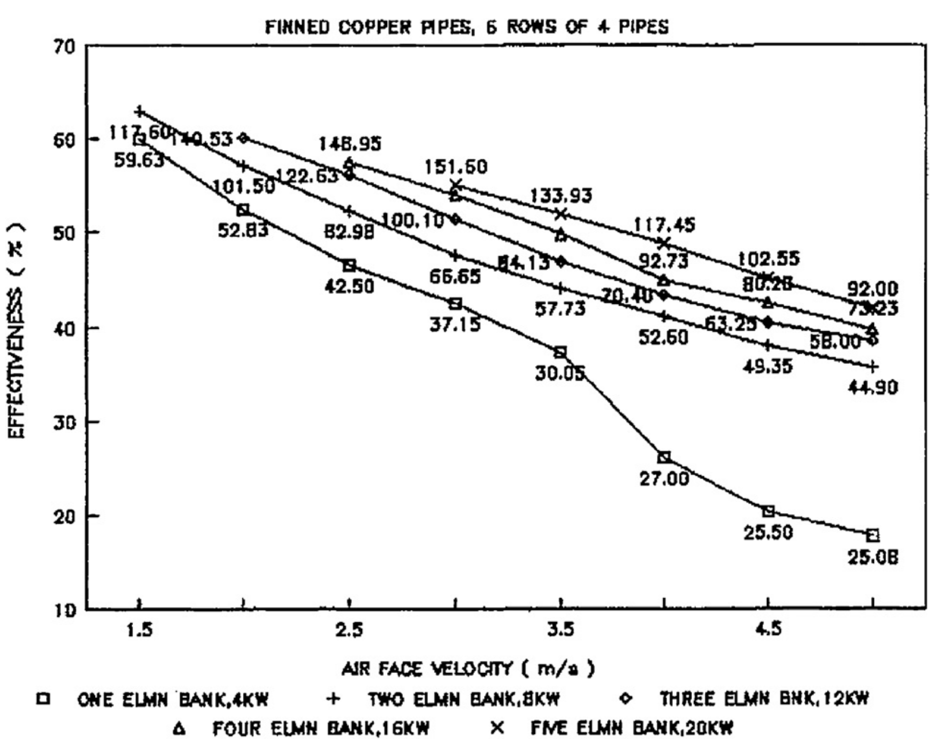


**Fig. 16.** Effect of air velocities flow through the evaporator section [5].

reliability. To support future research, goals to enhance the thermal performance of the heat pipes can be classified as follows:

(1) to modify the surface or the dimension of the heat pipes;
(2) to increase the time of the fluid flow over the evaporator and condenser sections;
(3) to arrange the tubes of the heat pipe appropriately such that the fluid contacts the evaporator and condenser section;
(4) to consider the types and qualities of working fluid and the volumetric filling ratio;
(5) to cause the working fluid to quickly bubble; and
(6) to cause the working fluid to flow in one direction for the CLOHP.

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